July 15, 2005

DECLARATION

The undersigned, Dana Scruggs, having an office at 8902B Otis Avenue, Suite 204B, Indianapolis, Indiana 46216, hereby states that she is well acquainted with both the English and German languages and that the attached is a true translation to the best of her knowledge and ability of PCT/DE 2004/001128 (INV.: HOEFS, R.).

The undersigned further declares that the above statement is true; and further, that this statement was made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code and that such willful false statements may jeopardize the validity of the application or document or any patent resulting therefrom.

Lana Scruggs

Dana Scruggs

185

10/542815 JC12 Rec'd PCT/PTC 20 JUL 2005

ELECTRICAL MACHINE

_
•
_
_

3

1

Background Information

4

The present invention relates to an electrical machine, in particular a generator for motor vehicles, according to the general class of the independent claim.

7 8

9

10

11

12

13

14

15

16

17

18

19

20

21

22

23

An electrical machine designed as an alternator is made known in DE 198 04 328 A1, in the case of which the generator shaft is supported by a movable bearing in the hub of a housing part. A spring element inserted in the hub exerts a load on the outer ring of the movable bearing with an axial force to achieve a defined rolling motion of the rolling elements in the movable bearing and thereby attain a longer service life of the movable bearing. The design of this bearing arrangement is relatively complex, since a number of components is required to exert an axial load on the roller bearing. In addition, every single generator and its bearing point must be adjusted to a defined initial axial load using a calibration step. Moreover, with a bearing arrangement of this type, only a relatively steep increase in force in the axial direction can be achieved. Due to longitudinal vibrations of the rotor, then, a relatively steep increase in spring force results and the load on the bearing therefore increases steeply. A steep increase in the spring load and the resultant high overall axial force load on the bearing is disadvantageous, since this could result in an overload, which would greatly shorten the service life of the bearing.

24

25

Advantages of the Invention

26

27

28

29

30

The electrical machine according to the present invention with the features of the main claim has the advantage that a large range of spring-force characteristics with a relatively flat increase in force is attainable. This means that the spring force and, as a result, the axial force load on the bearing may increase slightly

1 across the range of the intended compression of the spring element. As a result, 2 the service life that can be expected of this bearing is markedly increased. 3 4 Advantageous further developments of the electrical machine according to the 5 main claim are made possible by the measures listed in the subclaims. 6 7 Due to the fact that the spring element, in an outer region, bears against an outer 8 ring of a rolling bearing and, in an inner region, against a hub projection, this results - in combination with an inner ring of the movable bearing being secured 9 10 in a fixed manner on a rotor shaft – in a good application of load on this movable 11 bearing in terms of the criteria for the usual design of rolling bearings. 12 13 According to a further subclaim, it is provided that the hub projection has a 14 basically annular shape with a conical spring-support area that declines in the 15 outward direction. As a result of these features, the spring element bears in a defined manner in the radially inward region, which also makes it possible to 16 17 leave the effective leverage of the spring force unchanged or nearly unchanged. In addition, a simple tool geometry for manufacturing this surface against which 18 19 the spring element bears is made possible, thereby also extending the service life of the tool. Due to the fact that the spring element configured as a disc spring 20 21 essentially has the shape of a frustoconical shell, a high axial force can be achieved, so the rolling elements in the rolling bearing roll in their races under 22 23 defined conditions. This results in a favorable service life for the rolling bearing. 24 25 According to a further exemplary embodiment, it is provided that a spacer is 26 located in the force-transfer direction between the bearing and the spring 27 element. This spacer, as does the hub projection provided with an outwardly

declining conical spring-support surface, enables positioning in the hub without
causing a change in the leverage of the axially-acting spring force. The bearing is
therefore able to absorb axial vibrations relatively elastically without undergoing
excessive changes in force.

1	It can also be provided that a spacer is located in the force-transfer direction		
2	between the spring element and the hub, that fulfills the same purposes.		
3			
4	The spacer can be, e.g., an originally separate ring fastened to a spring element		
5	thereby resulting overall in an economical creation of a combination of spring		
6	element and spacer.		
7			
8	Drawing		
9			
10	Figure 1	shows a schematic illustration of a cross section through an	
11		electrical machine,	
12			
13	Figure 2a	shows a cross section through a bearing for supporting a rotor	
14		using a spring element,	
15			
16	Figure 2b	shows an enlarged depiction of a section of the hub projection,	
17			
18	Figures 3a		
19	and 3b	show two views of the spring element in Figure 2a,	
20			
21	Figures 4a	the state of the state of the state of the second of	
22	and 4b	show, in a highly schematic view, the behavior of the spring	
23		element in Figure 3a and Figure 3b when loaded by axial vibrations	
24		of the rotor,	
25	Ciarra Ca		
26	Figure 5a,	about three different views of a second exemplant embediment of a	
27	5b, and 5c	show three different views of a second exemplary embodiment of a	
28		spring element,	
29	Figure 6	shows a third exemplary embodiment of a spring element,	
30 31	Figure 6	Shows a third exemplary embodiment of a spring element,	
JΙ			

Figure 7 1 shows a cross section through the positioning sleeve in Figure 2a. 2 3 Description 4 5 Figure 1 shows, in a highly schematic view, a cross section through an electrical 6 machine 10. Electrical machine 10 has, among other things, two housing parts 7 13 and 16, that accommodate, among other things, a stator 19. Housing parts 13 8 and 16 each have a hub 21 that serves to support shaft 25 of a rotor 27 via a 9 bearing 23 and a bearing 24. 10 11 Left bearing 23 shown in Figure 1 is a "fixed bearing", and bearing 24 shown in 12 Figure 1 is a "movable bearing". Movable bearing 24 and its configuration and 13 position in hub 21 will be described in greater detail in conjunction with Figure 2. 14 15 Figure 2a shows, in a less schematic illustration, the arrangement of right bearing 16 24, movable bearing in hub 21, and bearing housing part 16. Housing part 16. 17 often also referred to as an end plate, has hub 21 in its center, the hub extending 18 axially in the shape of a cylindrical ring. A "hub projection" 30 adjoins hub 21, the 19 hub projection extending radially inwardly. Hub projection 30 is located on the 20 side of hub 21 facing away from rotor 27. Hub projection 30 has a basically 21 annular shape and is formed by a few spokes 32, among other things, oriented in 22 the radial direction. An annular spring-support surface 35 that is part of hub 23 projection 30 adjoins spokes 32 further radially inwardly. 24 25 A "fitting ring 38" made of plastic is inserted in hollow-cylindrical hub 21. Fitting 26 ring 38 serves to dampen vibration excitations between hub 21 and shaft 25. 27 Shaft 25 extends into fitting ring 38, the shaft being held in fitting ring 38 by 28 bearing 24. In the exemplary embodiment described, bearing 24 is designed as a 29 rolling bearing, and specifically as a roller bearing in this case. The roller bearing 30 is essentially composed of an inner ring 40, rolling elements 42 which are 31 spherical in this case, and outer ring 44.

An axially acting spring element 47 is inserted between bearing 24 and hub 1 2 projection 30. Spring element 47 is a disc spring that has an opening in its axial center that is provided for passage of shaft 25, which has a reduced inner 3 diameter here. Disc spring 47 essentially has the shape of a frustoconical shell 4 5 and, therefore, a radially inwardly directed inner region that bears against hub 6 projection 30 and spring-support surface 35. Spring element 47 bears against outer ring 44 of bearing 24 with a radially outwardly directed outer region. 7 8 Bearing 24, in turn, rests with its inner ring 40 on a shaft shoulder 50 of shaft 25. 9 10 To enable the most economical manufacture of the electrical machine possible, it 11 has been provided that, during construction, a relatively great tolerance between 12 spring-support surface 35 and left housing part 13 is permissible. It is also provided that the position of shaft shoulder 50 relative to spring-support surface 13 14 35 can be very different, so that, when bearing 24 bears against spring-support surface 35 via shaft shoulder 50 and spring element 47, very different, axially 15 16 acting spring forces acted between spring-support surface 35 and bearing 24 when conventional, known spring elements are used. This is not desired. Rather, 17 a range of spring-force characteristics with a relatively flat curve of force over 18 time in the tolerance range is provided. For this reason, it is provided that the 19 spring element configured as disc spring functions back and forth across a "flat" 20 21 position of the spring. The flat position of spring element 47 is defined to mean 22 that the outer region of spring element 47 has the same axial position as the inner region of spring element 47. This is synonymous with a shape of spring 23 24 element 47 that now exists, which is nearly a plane. 25 An electrical machine 10 is therefore provided, designed in particular as a 26 generator for motor vehicles, that has a rotatably supported rotor 27, whereby at 27 least one bearing 24 serves to support rotor 27 in hub 21, and an axially acting 28 29 spring force acts on bearing 24. Spring element 47 bears against hub 21 with spring force. Spring element 47 is a disc spring and is capable of functioning 30 back and forth across the flat position of spring element 47. 31

Spring element 47 bears, in an outer region, against outer ring 44 of bearing 24 1 configured as a rolling bearing and, in an inner region, against a hub projection 2 3 30. 4 Hub projection 30 and spring-support surface 35 are shown in an enlarged 5 depiction in Figure 2b. To better show the shape of spring-support surface 35, it 6 was shown in this highly exaggerated illustration unequivocally as an outwardly 7 declining, conical spring-support surface 35. The purpose of this highly conical 8 spring-support surface 35 is to prevent an over-proportionally great increase in 9 spring force between hub projection 30 and outer ring 44 during the axial 10 vibrations of rotor 27. It is provided that, in the extreme position of spring element 11 47, i.e., in a particularly highly deflected position position of spring element 47 in 12 the direction toward hub projection 30, spring element 47 does not come to rest 13 against a radially inwardly lying edge 53 of spring-support surface 35. If spring 14 element 47 were deflected to this extent, this would mean the spring force would 15 no longer bear against the radially outwardly directed side of spring-support 16 surface 35, but directly against edge 53, for example. As a result, the spring force 17 between outer ring 44 and hub projection 30 would increase abruptly and bearing 18 19 24 or spring element 47 might therefore be overloaded. 20 For the same reason, it is provided, among other things, in the exemplary 21 embodiment according to Figure 2a that, between outer ring 44 and the actual 22 spring part of spring element 47, an annular spacer 56 is secured between the 23 outer region of spring element 47 and outer ring 44. Without this spacer 56, 24 spring element 47 could bear, in a position in which it is pressed through to hub 25 projection 30, against an edge of the bore of inner ring 40. This would result in a 26 reduction of the effective leverage of the axial force here as well and, as a result, 27 the spring force would increase abruptly. To this end, spacer 56 is provided with 28 conicity. Spacer 56 essentially has the shape of a frustoconical shell, the larger 29

opening of which is directed toward spring element 47. The conicity is, e.g., 7°

(the cone angle is therefore 14°). Spacer 56 therefore bears radially outwardly

30

31

with a substantially narrow, circular ring-shaped surface against spring element 1 47 and can also be connected with spring element 47 via this circular ring-2 shaped surface or be secured to it, e.g., using an integrally joining connection 3 technique, such as welding, soldering or adhesive bonding. With this design, 4 spacer 56 avoids a shortening of the effective leverage in the region of the 5 transition from spring element 47 to spacer 56, which would occur with a flat 6 spacer 56 and a simultaneously greatly deflected spring element 47. Otherwise, 7 spring element 47 would bear against the radially inwardly located edge of 8 spacer 56 directed toward spring element 47, resulting in an increased load on 9 the spring element. A further reason for spacer 56 is that, due to spacer 56, it can 10 be ruled out that spring element 47 would bear against inner ring 40 in the loaded 11 and deflected state of spring element 47. Otherwise, there would be a risk that it 12 would wear through. A spacer 56 is therefore located in the force-transfer 13 direction between bearing 24 and spring element 47. 14 15 A view of spring element 47 from the left is shown in Figure 3a, based on Figure 16 2a. The view of spring element 47 shown in Figure 3b is a view from the right. 17 The annular shape of spacer 56 and spring element 47 is clearly recognizable. 18 Spring element 47 has an overall wave-shaped inner region. In this example, a 19 total of three pegs 59 are formed on the outer edge of spacer 56, the function of 20 which will be explained below in conjunction with Figure 7. The number of pegs 21 59 can also deviate from this; for instance, four or six pegs are also possible. 22 23 A schematic view of spring element 47 is shown in Figure 4a and Figure 4b. In 24 both cases, the axial deflections of spring element 47 are shown in a greatly 25 exaggerated manner to clearly illustrate the changes. Figure 4a illustrates the 26 force and position relationships of a first operating state, in which a supporting 27 force F₁ between spring-support surface 35 and spring element 47 acts on the 28 radially inner side of spring element 47. An equal force F₁ acts between outer 29 ring 44 and, in this case, spacer 56. Line I₀ was selected as the reference line. 30 which, in this case, lies in the plane of osculation between spring element 47 and 31

spring-support surface 35. The axial height of spring element 47 between the 1 side of spring element 47 facing hub projection 30 and the plane of osculation is 2 d₁. The further extreme case in Figure 4b shows a spring element 47 deflected 3 fully to the right past the flat position, whereby the side of spring element 47 4 facing hub projection 30 - the contact surface between outer ring 44 and spacer 5 56 - is now on the other side of flat position 10, i.e., d2. It is also clearly shown 6 that force F₂, which is now acting between outer ring 44 and spring element 47, 7 and between spring element 47 and spring-support surface 35 is nearly as great 8 as force F₁ that was applied previously. 9 10 A further exemplary embodiment of a spring element 47 is shown in three 11 different views in Figure 5a, Figure 5b and Figure 5c. Similar to Figure 3a, the 12 illustration in Figure 5a is also a view from the left; accordingly, the view in Figure 13 5c is a view from the right, similar to the illustration in Figure 3b. As is the case 14 with the exemplary embodiment of spring element 47 described above, this 15 exemplary embodiment also has a radial inner region that has a generally wave-16 shaped configuration. The outer region of spring element 47 is slotted in close 17 intervals and has alternating radial positioning elements 62 that serve to 18 concentrically position spring element 47 in the fitting ring 38 and, therefore, to 19 concentrically position the rotor axis. As an alternative, indirect concentric 20 positioning can take place between spring element 47 and hub 21. Spacers 56 21 are located between the positioning elements, which bear against outer ring 44, 22 as they do in the previous exemplary embodiments. Overall, these spacers 56 23 are curved and are U-bent at a nearly right angle. 24 25 Figure 6 shows a third exemplary embodiment of a spring element 47. It is 26 provided that spring element 47 and spacer 56 for outer ring 44 are configured as 27 a single component. In addition, but which can also be used individually, a further 28 spacer 56 is located in the force-transfer direction between spring element 47 29 and hub 21, which is also integrally joined as a single component in this case. 30

31

Finally, a sectional view of fitting ring 38 is shown in Figure 7. Fitting ring 38 has 1 insertion pegs 70 on one axial end, which are pressed between spokes 32 of 2 housing part 16 and, in addition, as also shown in Figure 2a, are configured as 3 snap-in elements and engage in grooves between spokes 32. Sections between 4 pegs 70 also interact with spokes 32 such that spokes 32 acts as stops for fitting 5 ring 38. Pegs 70 limit the play of fitting ring 38 in the axial direction toward the 6 left, and the intermediate spaces between pegs 70 limit play toward the right, 7 thereby resulting, overall, in a defined position of fitting ring 38 in hub 21. In 8 addition, three grooves 73 are formed on the cylindrical inner circumference of 9 fitting ring 38, into which pegs 59, mentioned above (refer also to Figures 3a and 10 3b) are inserted, thereby resulting in an overall unequivocal position of spring 11 element 47, according to the exemplary embodiment in Figure 2a, 3a and 3b, in 12 fitting ring 38. An incorrect position of spring element 47 in hub 21 is thereby 13 ruled out. For the case in which an unequivocal match between spring element 14 47 and hub 21 is not required, pegs 59 and grooves 73 can also be distributed 15 evenly around the circumference. 16 17 18 19

20